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The design and construction of an inertia welder

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THE DESIGN AND CONSTRUCTION
OF AN INERTIA WELDER

BY

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THESIS

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ABSTRACT

The purpose of this investigation was the design and fabrication of a machine to bond metal to metal using the inertia method of friction welding. This method of bonding is considered far superior to the heretofore employed friction welding processes as will be explained later. The inertia method controls the energy imparted to the weld by using a flywheel which has been brought to a state of predetermined kinetic energy due to rotation. This energy is then transferred to the weld zone in the form of heat due to rotational friction.

Hollow specimens with a cross-sectional area of 0.306 square inches have been successfully welded on the present machine. The results show that the machine as designed would be capable of welding materials 1 inch in diameter were it not for the flywheel velocity being limited to 1,000 revolutions per minute due to unbalance.

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I. INTRODUCTION

Friction welding is a welding process in which the energy is supplied by a motor and delivered directly to the weld zone. This method requires large motors to prevent stalling when the two parts are brought together under pressure and the rate of energy transfer is restricted by the size of the motor used. Due to the relatively slow rate of energy transfer, the time necessary to build up heat at the weld zone is sufficient to allow much of the heat produced to be conducted away from the weld zone by the adjacent material. This requires more energy than should be needed and, for example, makes welding of rod to flat plate very difficult due to the large area of the plate acting as a heat sink. Also, after the weld zone is heated to a plastic state, it is necessary to stop rotation which requires brakes or reversing of the motor or both. It can be readily understood that, as larger motors are needed, the braking requirement also increases. This method of friction welding, even when automated, rarely uses more than 50 percent of the energy expended in actually making the weld, due to heat loss to surrounding material and energy required to stop the process. Because of the large volume of material to be heated and the length of time during which heating occurs, cooling of the weld zone is relatively slow, allowing grain growth and oxidation to impair the quality of the weld.

The inertia method uses a smaller motor to bring a flywheel to a desired level of energy after which the flywheel, disconnected from its power source, transfers its energy to the weld zone quite rapidly as the parts are brought together under pressure. This energy transfer is rapid and the heat sink effect does not occur. Since the weld cycle is complete when the flywheel stops after transferring all of its energy to the weld zone, no braking is required. This means that approximately 95 percent of the energy required is used directly in making the weld. Also, the relatively small volume of material heats quickly, cools quickly, and prohibits grain growth and oxide formation thus giving a stronger weld.

II. REVIEW OF LITERATURE

In recent years most metal bonding processes which contributed significantly to commercial application were either pressure, fusion, or brazing processes. These processes have not filled all needs since they do not give high-strength bonds under certain conditions. For example, arc welding leaves a course-grained structure in the weld zone which detracts from the mechanical properties of the weld and cannot be used to join metals whose atomic diameters vary greatly.

In the inertia welding process, the mechanical energy of the flywheel is rapidly converted to heat energy at the weld interface due to friction. When transfer is complete, the flywheel comes to rest and the cycle is complete. Figure 1 shows the temperatures achieved and the width of the heat-affected zones for friction welding and inertia welding. The complete inertia welding cycle takes from 0.2 seconds to 3.0 seconds as compared with 20.0 seconds to 40.0 seconds, or more, for friction welding. The rapid heat build up in the inertia welding process, and subsequent rapid cooling, leaves a fine-grained structure. Figure 2 shows the optimum and permissible speeds in surface feet per minute for solid bars 1/4 inch to 4 inches in diameter. Tubular parts can be run at slower speeds. The pressure applied, and the rotation of one part against the other, forces some of the plastic material out of the weld zone as flash. This flash carries any surface contamination with it and leaves a weld zone with very

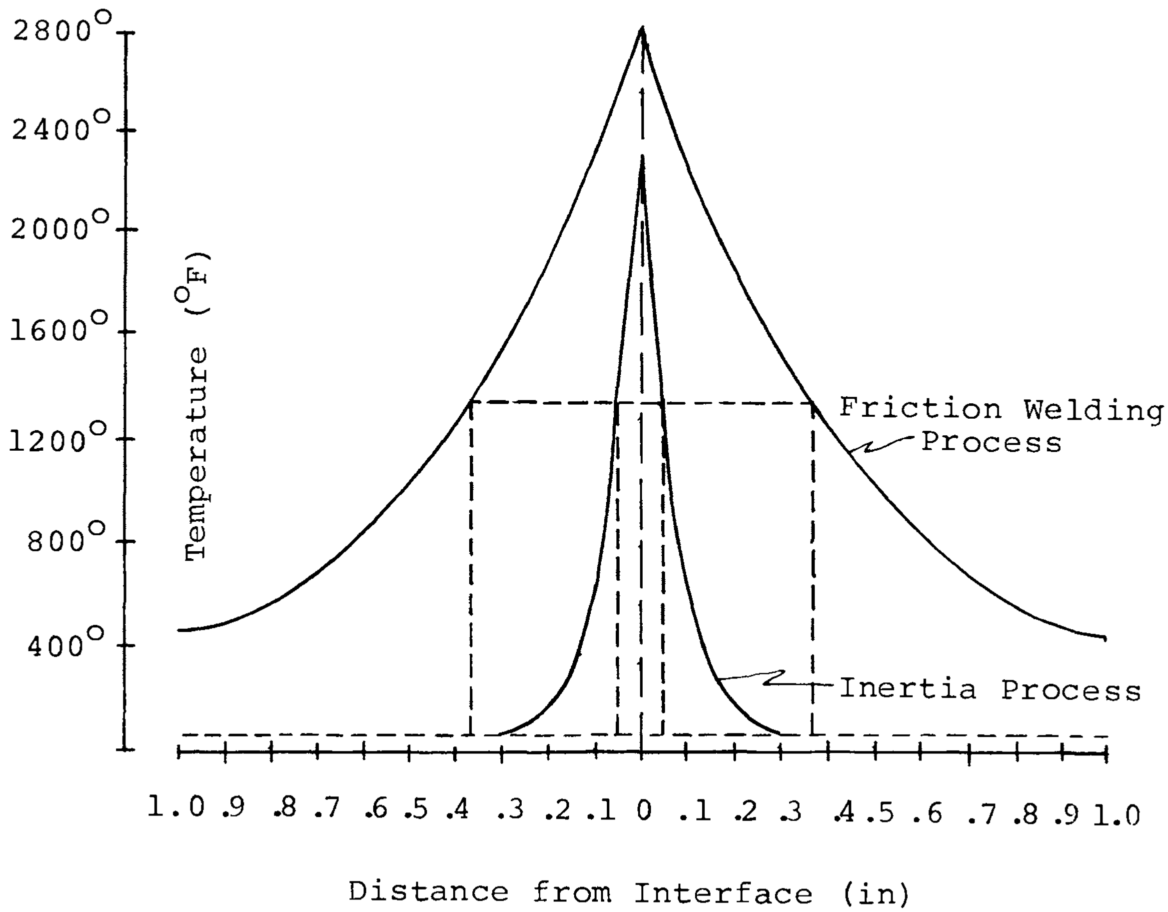


Figure 1. Temperatures and Heat Zones

Courtesy of Caterpillar Tractor Company
Peoria, Illinois

few inclusions in most cases. This gives excellent mechanical properties without the need for heat treating. Figure 3 shows the three stages of the weld cycle. During the third stage, which is approximately the last 1/3 to 1/2 revolution of the flywheel, the bonded material is plastically worked. This plastic working is a process of slip occurring between planes of atoms reorienting them and giving a stronger and tougher structure. This plastic working also leaves flow lines that run parallel, or tangent to, the outer edge of the part rather than intersecting it normally as in the upset stage of friction welding.

The parameters affecting inertia welding are:

1. Flywheel mass & configuration
2. Flywheel angular velocity
3. Pressure applied

The Caterpillar Tractor Company, in their investigations, used flywheels up to 2,000 pounds, angular velocities up to 12,000 revolutions per minute, and forces as high as 500,000 pounds.

Each of the three parameters mentioned has its own significant effects on the operation and the finished weld.

The use of inertial weights has the advantage of lower power requirements since the flywheel is used to store, after accumulation, the energy required for the weld. The ability to change weights makes for ease of adjusting energy

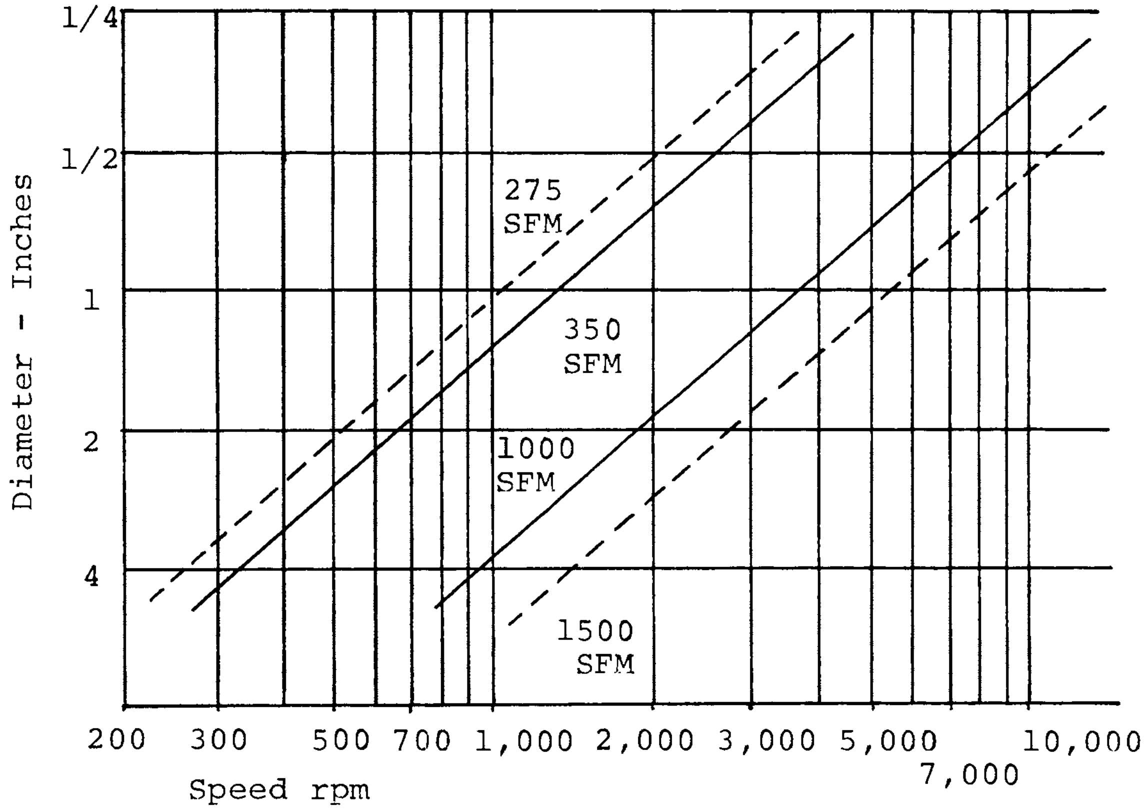


Figure 2. Optimum and Permissible Speeds

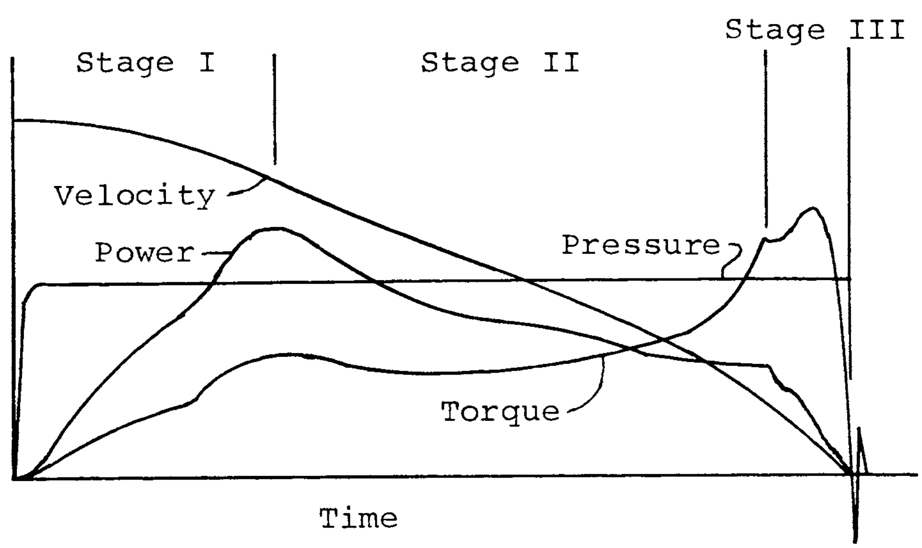


Figure 3. Stages of Welding Cycle

Ibid

requirements to achieve more uniform results. The amount of plastic working of the bonded material can also be regulated in part by the use of inertial weights.

The amount of pressure applied can be varied to achieve desired results. In general, bonding pressures used are approximately one-half of the yield-point strength of the materials bonded; however, pressures nearer the yield-point strength produce bonds with narrower heat-affected zones in shorter cycle times. Pressures very near the yield strength can reduce the temperature required at the interface to give a high-strength bond. This is advantageous when bonding materials with high thermal conductivity where it is difficult to retain the high temperatures required for bonding with lower pressures. Higher pressures are also advantageous when bonding dissimilar materials where the melting temperatures are different, and it would be difficult to obtain high enough temperatures in the higher melting point material with lower operating pressures.

The angular velocity is readily varied either by pulley changes or the use of a Variac and a variable speed motor. The variation of angular velocity is advantageous in varying the energy stored in the flywheel while the flywheel size is held constant. It is also necessary to use different angular velocities for different materials or sizes of materials to be bonded. Angular velocity is used in conjunction with flywheel mass inertia in setting the energy level of the flywheel prior to the start of the welding cycle.

The energy stored in the flywheel is determined by the formula:

$$E_k = \frac{I_o \omega^2}{2}$$

where

E_k = kinetic energy of rotation - ft. lbs.

ω = angular velocity - radians per second

I_o = moment of inertia - lb. ft. sec.^{2}

$= \frac{WR^2}{2g}$ (for a right circular cylinder)

where

W = weight - pounds

g = acceleration of gravity - feet per second^2

R = radius of weight - ft.

The energy imparted to the weld is in the range of 15 to 50 horsepower per square inch of weld cross-sectional area, one horsepower being equal to 550 foot-pounds per second. This rapid energy transfer cannot be achieved using standard friction welding. The effects on the weld zone of varying the three parameters are shown in Figure 4.

A. Metallurgy

It is necessary to consider the following points concerning metallurgy when making a weld:

1. Oxide formation
2. Non-metallic inclusions
3. Effects of heat
4. Cooling rate
5. Plastic working

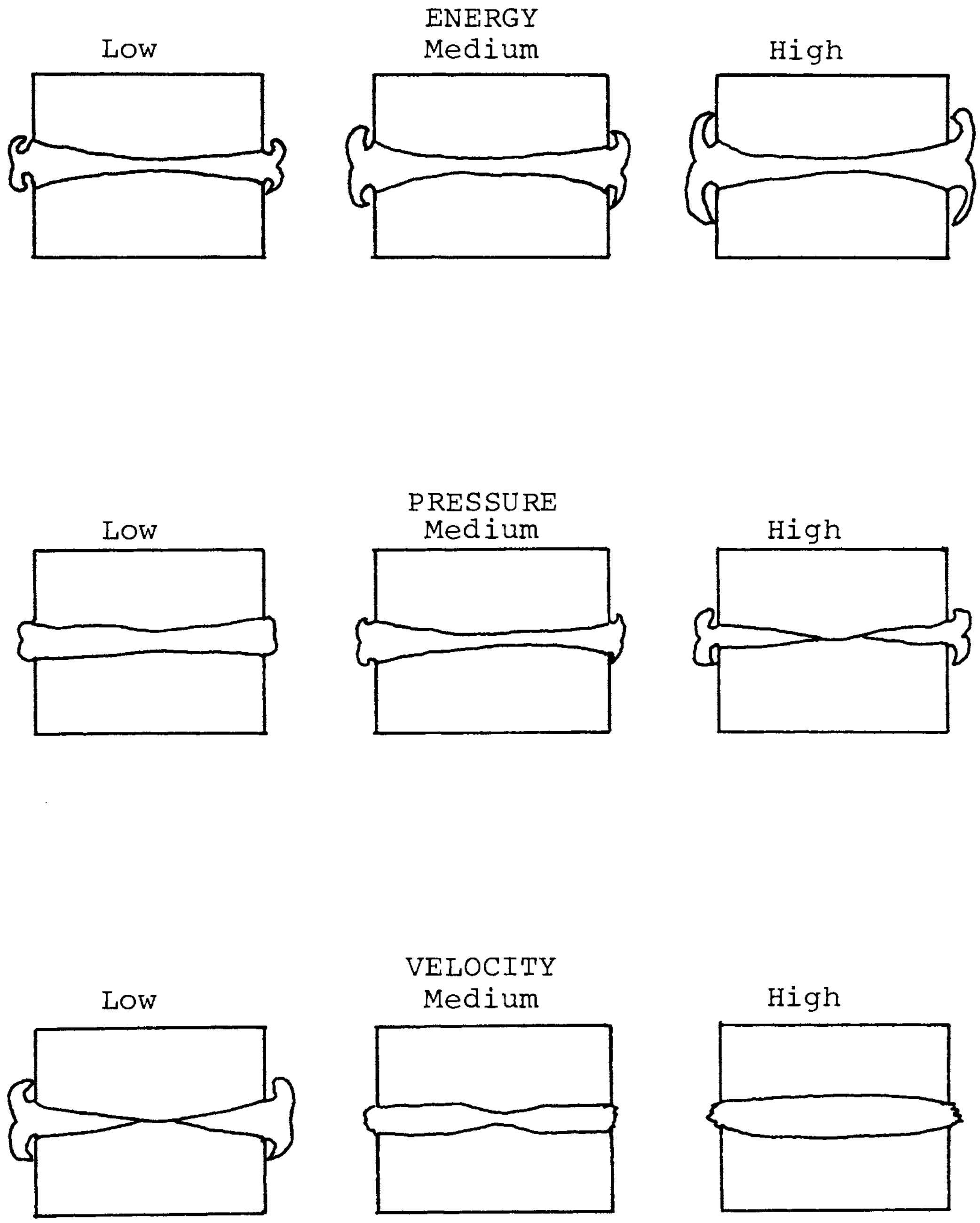


Figure 4. Effects of Parameter Variation

Ibid

The formation of metallic oxides occurs when most metals are in contact with air, and the rate of oxide formation increases with increasing temperature. The inertia process builds up heat at the interface so rapidly that little volume of material is heated, and cooling is quite rapid after the weld is complete. Therefore, the time during which oxide formation would be the most rapid is very limited. Oxides that may have formed on the interface prior to welding are expelled from the weld zone with the flash and do not affect the strength of the finished weld.

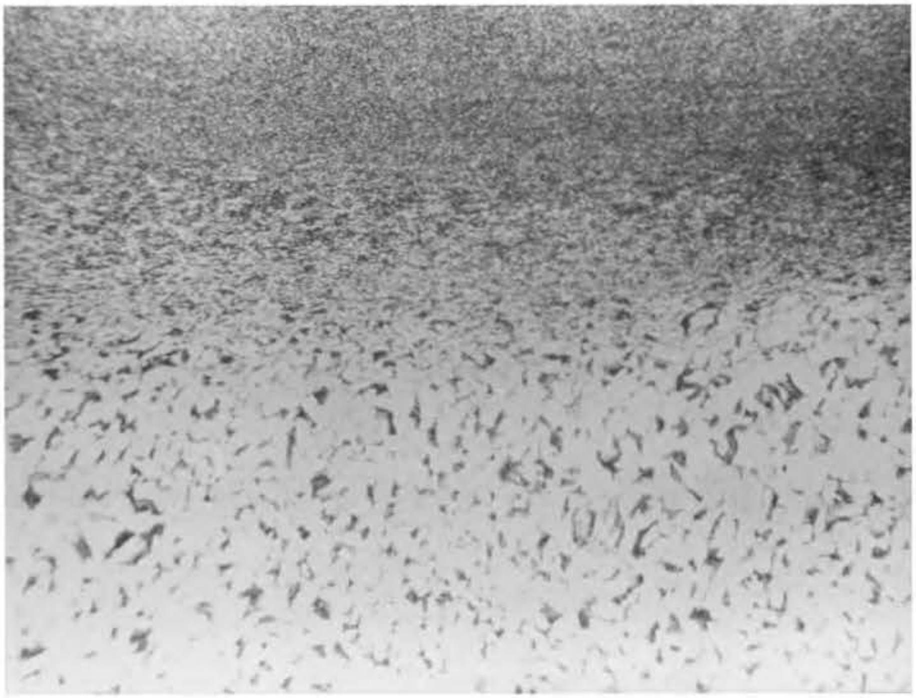
Non-metallic substances, such as cutting oil, dirt, slag, or other foreign substances which might be located at the interface, are also usually expelled with the flash. If this inclusion were in the metal itself, not all would be expelled. The remainder would be distributed throughout the weld zone by plastic working during the third stage so that its effect would be minimized.

Heating of metals, for example carbon steel, brings about the possibility of physical changes taking place in the area of heat application. Since the rate of heat build up is rapid, as is the rate of cooling, a very fine-grained and hardened structure is accomplished in the weld zone. This gives added strength, toughness, and corrosion resistance to the weld zone.

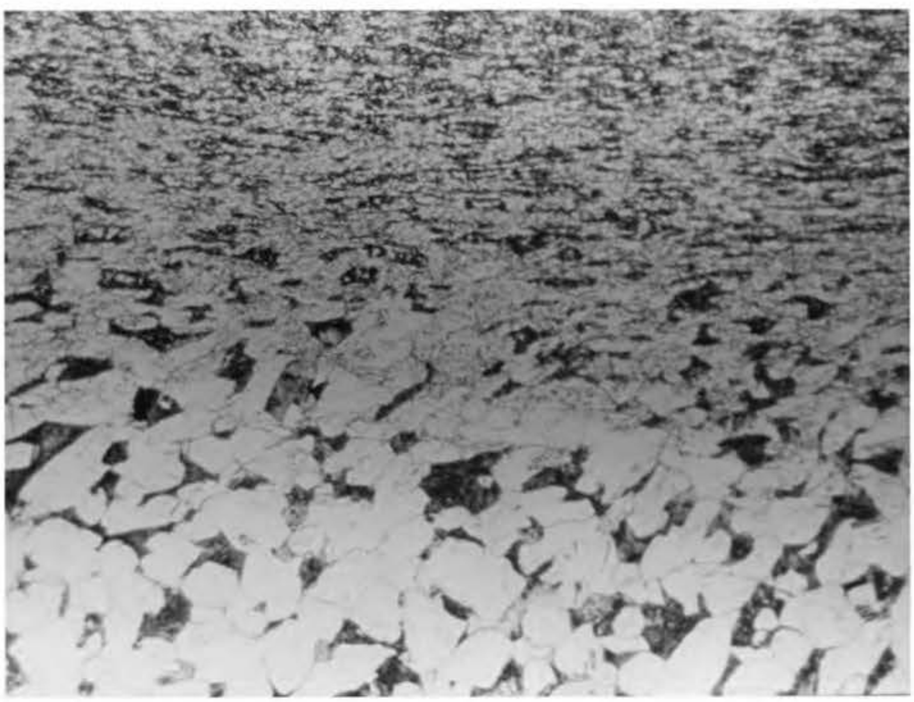
Plastic working during the third stage, after the bond is formed, has a work hardening effect as well as refining grain size and fragmenting and mixing inclusions throughout the weld zone. It also tends to fill in any voids that might be present due primarily to the applied pressure. This working also mixes metal from each part into an alloy, more noticeable with dissimilar metals, to give a strong homogeneous bond.

As a specimen is welded, the following metallurgical changes take place:

1. Heat is generated at the interface due to friction. Relative motion and axial force accomplish plastic deformation of the heated interface material.
2. Plastic deformation tends to elongate the grains of the material in the direction of motion.
3. The high temperature present at the interface allows a recrystallization of the plastically deformed material.
4. After the flywheel comes to rest, the heated mass cools relatively quickly, preventing severe grain growth from occurring after the recrystallization.
5. The photographs taken at 100X, 250X, and 500X show the progressively smaller grain size as the weld interface is approached and there is no apparent interface, as can be seen in Figures 5 and 6.



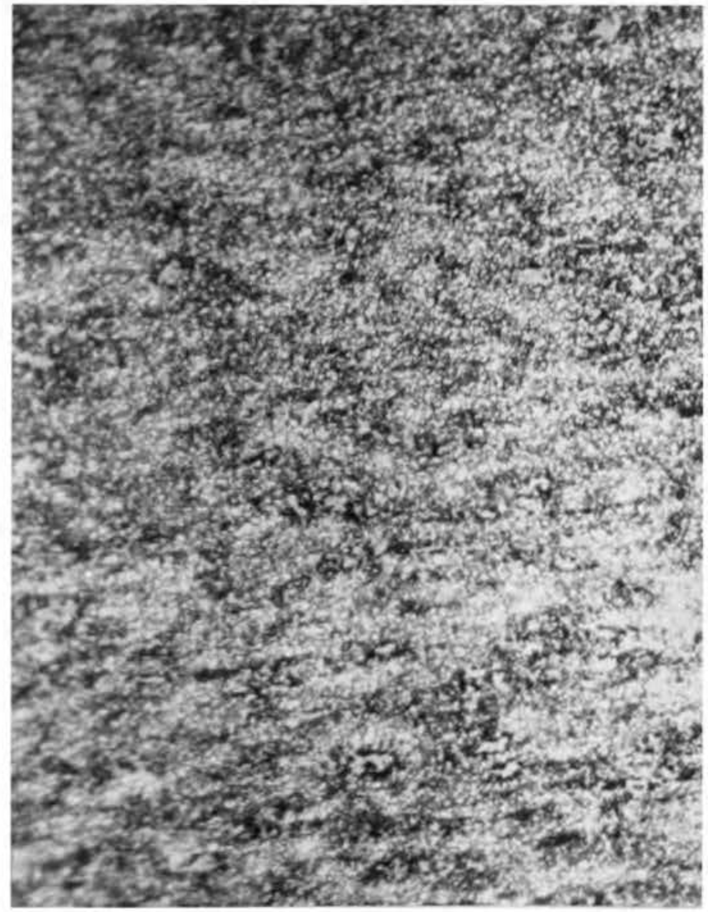
Magnification 100X



Magnification 250X

Figure 5. Microstructure of Weld Area

Magnification 500X



Magnification 500X
Weld Zone

Figure 6. Microstructure of Weld Area

B. Welding Criteria

The welding, or joining of metals, brings into consideration the following questions:

1. How is the heat supplied?

   In this investigation, heat is supplied by friction obtained from a combination of flywheel energy and axially applied pressure.

2. What temperatures are required?

   For this type of solid-state welding of SAE 1020 steel, a temperature of 2300° F. to 2400° F. is required.

3. What changes occur in the properties of the metal?

   The heat-affected zone is quite narrow due to the speed of welding and the lower temperature involved. There is some grain refining and hardening near the interface of the weld.

4. Can these changes in properties be controlled?

   The changes that occur can be controlled so that each successive weld is identical to the previous weld within close tolerances.

5. Is a flux required?

   No flux is required. Any impurities or surface contamination at the weld interface is forced out of the weld zone, due to rotation and pressure, and forms a flash or upset type of residue around the outside of the weldment. This can, if required, be machined off.

6. Is pressure needed?

   Pressure is needed in order to produce friction which, with the flywheel inertia, generates heat to assist welding. It also aids in ejecting impurities from the weld area and bringing fresh parent material in contact for good bonding.

7. Is the process adaptable to mass production?

   The use of automating controls makes for ease of operation, and the fast cycle time makes this method quite adaptable to mass production.

8. Is the cost compatible with other methods?

   Although no estimates of cost of production were made in this study, the saving in labor time for a given production quota, plus the saving on post-welding operations should make inertia welding quite attractive on a cost-of-production basis.

C. Design

It was necessary initially to determine the amount of heat required to bring about a plastic condition at the weld interface. Very little has been written on friction welding, especially on the heating due to friction, since the coefficient of friction is continually changing in value. This continual change can be explained as follows: as the parts are brought together, they are at room temperature and the interface has some degree of roughness due to whatever method of machining is used. The temperature and surface finish of the specimen, together with the load applied and the angular velocity of the specimen, gives a certain value for the coefficient of friction. Some of the surface roughness is immediately ground off and/or flattened by the pressure exerted, which by itself would change the value of the coefficient, even though temperature and angular velocity might remain constant. However, the angular velocity must be continually decreasing due to the friction and the fact that no additional energy is being added to compensate for this energy transfer. Also, friction has caused heating of the interface material which affects the value of the coefficient. Since all parameters, except the pressure exerted, are continually changing from start to finish of a cycle, the value of the coefficient must also continually change. Due to this, it was necessary to make the following assumptions:

1. The amount of heat required is proportional to the cross-sectional area of the weld.
2. The pressure required is proportional to the cross-sectional area of the weld and must be less than

that which would exceed the yield point stress for the material.

3. The flywheel size necessary to store sufficient energy to produce the required head due to friction is proportional to the cross-sectional area of the weld.

The fact that the Caterpillar Company, who originally built an inertia welder, used a 2,000 pound flywheel and 200,000 pounds of axial force to weld a 4-inch diameter bar was the basis for design. A 4-inch diameter bar has a cross-sectional area of 12.56 square inches, whereas a 1-inch diameter bar has a cross-sectional area of 0.785 square inches. Therefore, the 1-inch diameter bar has 1/16 the cross-sectional area of the 4-inch diameter bar. Using this ratio and the afore stated assumptions, the following design criteria were found:

Flywheel weight $W = 2{,}000$ pounds $\div 16 = 125$ pounds

Axial force $F = 200{,}000$ pounds $\div 16 = 12{,}500$ pounds

The flywheel size is also related to moment of inertia and radius of gyration; in this study weight alone was used for design. Since these values were based on assumptions, a safety factor was necessary. The hydraulic cylinder that was available for use had a capacity of 16,000 pounds, which was 28 percent above the assumed value of 12,500 pounds. The flywheel weight was selected to be 200 pounds. With these criteria, the machine was then designed.

The four main columns of the machine each sustain 1/4 of the 16,000 pounds of axial thrust or 4,000 pounds. A 1-inch diameter bar (0.785 square inch cross-sectional area) would have a tensile force of 4,000 pounds ÷ 0.785 square inches or 5,100 pounds per square inch, which is well below the yield point stress of 40,000 psi for low quality 1020 carbon steel. Due to the fact that the material available was of unknown composition, and because of the required amount of steel bar, 2-inch diameter stock was used for the columns. The columns were cut to 32 inches in length and both ends were turned to 1.750 inches for 2.500 inches in order to fit through the top and bottom plates. Both ends were then threaded to accept 1 1/2-12 NF semi-finished hex nuts. Flat washers and split-ring lock washers were used under the nuts. The stress the columns will actually be under is dependent on the stress area of the threads. For a 1 1/2-12 NF thread, the stress area is 1.5799 square inches; therefore, the actual stress in the weakest part of the column would be 4,000 pounds ÷ 1.5799 square inches, or 2,530 psi.

The top plate was machined from a piece of 1 1/4-inch thick steel plate which, due to a badly rusted and pitted surface, was milled to a thickness of 1 1/16 inches and 16 inches square. A center hole 2 1/4 inches in diameter was machined to accept the hydraulic cylinder ram. Four holes, 11/16 inch in diameter, were drilled 90° apart at a radius from the center of 2.31 inches to accept the longer tie-rods necessary to affix the cylinder to the top plate. All five of these holes were slightly larger than needed to allow alignment of the cylinder axially with the flywheel shaft during assembly. Four 1 3/4-inch diameter holes were then drilled at the radius of 8 inches from the center to accept the main columns. All but the center hole were centered on the diagonals drawn between opposite corners of the plate. The plate was then set up in a testing machine under loading conditions similar to those it would sustain under maximum conditions of actual use. The following loads and deflections were noted:

Load In Pounds	Deflection In Inches
5,000	0.0110
10,000	0.0150
15,000	0.0185
20,000	0.0220

These deflections were taken near the center hole and were less than 1/32 inch. On releasing the load, the plate returned to zero deflection, which indicated the yield point of the material had not been exceeded. Since this loading was 25 percent in excess of maximum loading under actual conditions, the plate was acceptable for use in the machine.

The base plate was machined from 1-inch thick steel plate, and its use based on the test made on the top plate. It proved adequate under maximum loading. It was machined 16 inches square and four holes drilled for the main columns as with the top plate. A hole, 2.375 inches in diameter, was drilled and bored through the center of the plate to allow the rear of the bearing shaft to protrude. Four holes were drilled and tapped at a radius of 2.750 inches from the center and 90° apart to accept 5/16-24 NF hex head cap screws which were used to secure the bearing cover to the base plate.

The bearing cover was machined from 1-inch thick steel plate. A center hole was drilled and bored to 2.720 inches in diameter. It was then turned to a 4 1/2-inch diameter, leaving a 1-inch flange 5/16-inch thick all around. The cover was then counter-bored from the flange side to a diameter of 3.935 inches and a depth of 0.827 inches to receive a new departure bearing #3211. Four 11/32-inch diameter holes 90° apart at a radius from the center of 5 1/2 inches were drilled to allow mounting to the bottom plate, using 5/16-24 NF hex head cap screws along with flat washers and split-ring lock washers. The stress in the four 5/16-24 NF threads was the limiting factor for the bearing cover and bearing. The total load supported by the bearing is one-half of the flywheel, flywheel shaft, and chuck and is less than 150 pounds. The load in each thread is 150 pounds/4 threads = 37 1/2 pounds. The stress is 37 1/2 pounds ÷ .058 square inches = 646.55 psi. The thread is capable of carrying 40,000 psi ÷ .058 square inches = 2,620 pounds.

The bearing shaft was made from 3-inch diameter steel and was turned to 2.750 inches in diameter. The one end was then turned to a diameter of 2.167 inches to conform to the bore of the #3211 bearing, for a distance of 2.000 inches.

The center portion, for a distance of 1.125 inches, was left as previously turned and the remaining 2.000 inches was turned to a diameter of 1.375 inches to press fit into the flywheel shaft, after the flywheel base plate was attached. Loading was too small to require calculation.

The flywheel base plate was made from 1 1/8-inch thick steel plate. A center hole was threaded 2 1/2-12 NF to accept the flywheel shaft and the rear of this threaded hole was counter-bored to a diameter of 5.248 inches and a depth of 0.500 inches to accommodate a #T-302 Timken thrust bearing, press-fit. The outside diameter of the plate was turned to 11.875 inches and four holes were threaded 1/2-20 NF, 90° apart at a radius of 5.500 inches to accommodate the flywheel rods.

The flywheel rods were 5/8-inch diameter steel, 8 inches long. Each end was threaded 1/2-20 NF, for a distance of 1 inch. Since the method of crossing each successive pair of flywheel plates, Figure 7, as the flywheel was built up, would prohibit their movement due to the interference of the flywheel shaft, it was necessary only to determine if the flywheel rods would deflect sufficiently to allow the plates to slip past the flywheel shaft and exert a high shear stress on the rods. Assuming that each rod acts as a cantilever beam, the loading was determined as follows:

$$F_c = \frac{W r_o \omega^2}{g}$$

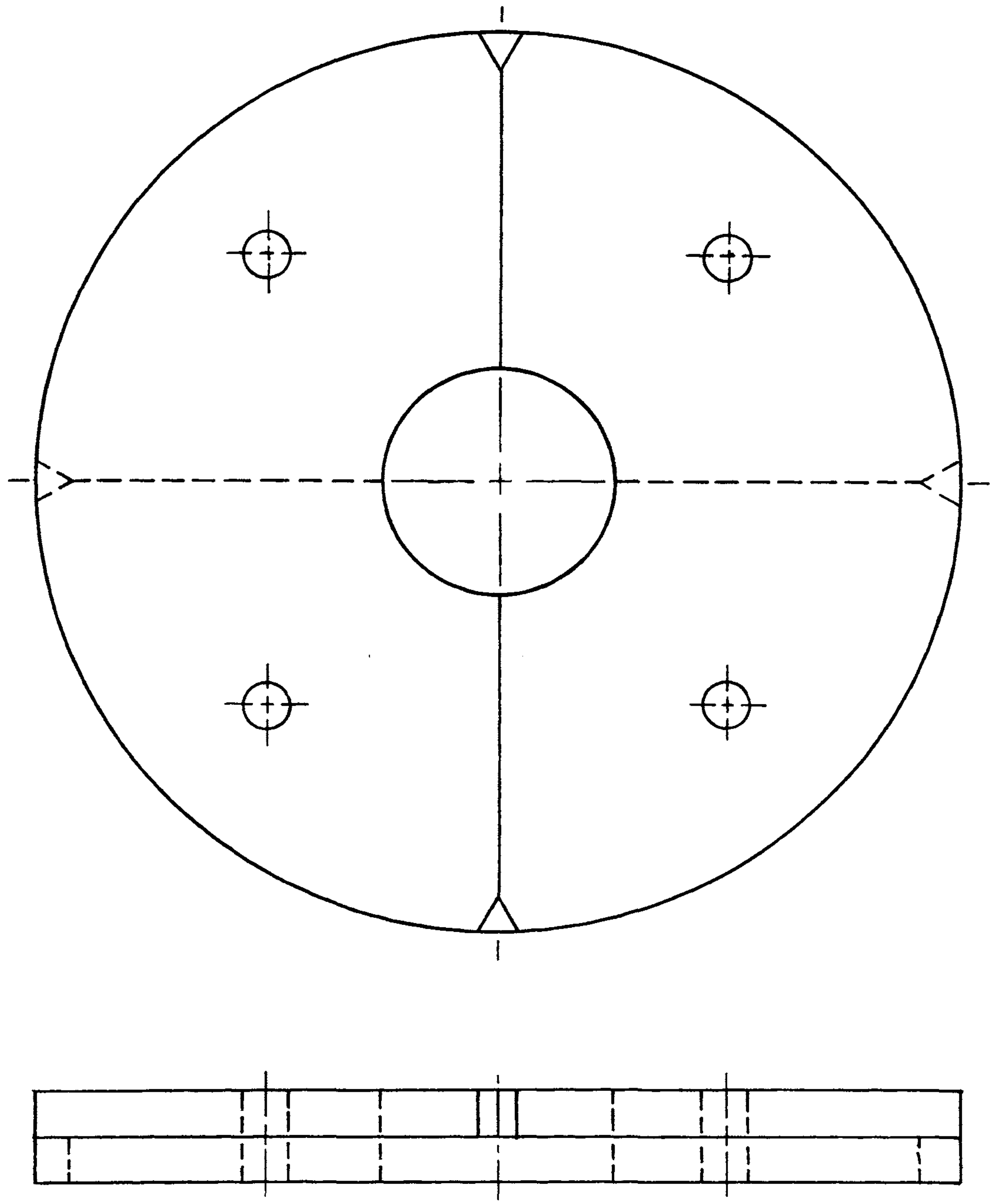


Figure 7. Flywheel Plate Set-Up
(Not to Scale)

where

F_c = force in pounds

W = weight in pounds

g = acceleration of gravity in inches per second^2

ω = angular velocity in radians per second

r_o = radius to center of gravity in inches

$= 3/16\pi\,(\frac{R^4-r^4}{R^3-r^3})$

R = outer radius of half shell in inches

r = inner radius of half shell in inches

therefore

$$r_o = 3/16(3.14)(\frac{1296-5}{216-3}) = 3.57 \text{ inches}$$

and

$$F_c = \frac{15(3.57)(183)^2}{386} = 4{,}660 \text{ pounds}$$

or 2,330 pounds on one rod. The next plate gives an equal force but at 90° to the first which, when vectorally added, gives a resultant force for the two plates of 3,300 pounds outward from the center axis. There are six such loads when the entire flywheel is in use. The deflection due to this load is

$$y_{max} = WL^3 \div 8EI$$

where

y_{max} = maximum deflection in inches

W = total load in pounds

L = total length of beam in inches

E = modulus of elasticity

I = moment of inertia

$$= \pi D^4 \div 64$$

where

D = diameter in inches

therefore

$$y_{max} = \frac{19,800(6)^4}{8(30 \times 10^6)(0.153)} = 0.117 \text{ inches}$$

This deflection is not sufficient to allow the plates to slip past the flywheel shaft and bring force to bear on the rods that might cause failure.

The flywheel shaft was made from 3-inch diameter steel rod. One end was turned and threaded 2 1/2-12 NF for 5/8 inches, to screw into the flywheel base plate. A portion 7.750 inches long was left at a diameter of 3.000 inches, and the 2.156 inches remaining were turned to a diameter of 2.167 to accommodate a second #3211 bearing. The other end was turned and threaded 2 1/4-8 Pitch for 1.500 inches to accept the flywheel chuck. Compressive loading and flywheel weight are insignificant in this size shaft and need no stress calculations.

The flywheel shaft, flywheel base plate, bearing shaft, and thrust bearing were then assembled.

The bearing plate was made from 1/2-inch thick steel plate. It was milled to 16 inches square and four 2-inch diameter holes were drilled to accept the four main columns as with the top and bottom plates. A center hole was bored 3.935 inches in diameter to accommodate a #3211 bearing.

Four holes, two each on opposite sides, located 1 3/8 inches from the ends were drilled and tapped 1/2-20 NC for cup point set screws to lock the plate to the main columns. Four holes at a radius of 2 1/2 inches from the center, 90° apart, were drilled and tapped 5/16-24 NF to accommodate the bearing retainer.

The bearing retainer was made from 1/8-inch thick aluminum plate. It was turned to a 6-inch diameter and a 2 23/32-inch diameter hole was bored in the center. Four holes, 11/32-inch in diameter, 90° apart were drilled at a radius from the center of 2 1/2 inches. The retainer was then fastened to the bearing plate using 5/16-24 NF hex head cap screws, 5/16 washers, and 5/16 split-ring lock washers.

The ram adapter was made from 3-inch diameter steel rod. The right end was turned to a diameter of 2 1/8 inches for a distance of 1 1/8 inches. The center portion, 7/8 inch wide, was turned to a diameter of 2 inches. The remaining 1 1/8 inches was turned and threaded 2 1/4-8 Pitch to accommodate one of the chucks. The other end was drilled and threaded 1-14 NC for 1 5/8 inches to allow the adaptor to be screwed onto the cylinder ram. This part was later changed in the following manner. A hole, 1 inch in diameter, was drilled in the center of the chuck end to a depth of 2 inches. Four holes, 90° apart and 1 inch from the end, were drilled and tapped 3/8-16 NC to receive hex socket, full dog, set screws. This hole and set screw set up was then used as one specimen holder in lieu of a chuck.

The ram brace was made from 1/4-inch thick steel plate, 5/8-inch diameter steel rod, and 1-inch diameter steel rod. Since this is an assembly, the brace components and assembly are shown in Figure 8.

The flywheel plates were made from 1/2-inch thick steel plate. There were 12 required (24 half plates). 12 inch diameter circles were marked off on steel plate and were then torch cut slightly oversize. A 2-inch diameter hole was drilled in the center of each for mounting on a lathe chuck. The outside was turned to 11 7/8-inch diameter. The plate was remounted and the center hole enlarged to a 3-inch diameter to match the flywheel shaft. Four holes, 90° apart and 21/32 inch in diameter, were drilled at a radius from the center of 5 1/2 inches to accommodate the flywheel rods. The plates were then sawed in half along a line midway between the flywheel rod holes so they could be mounted without partial disassembly of the machine. Finally, a small triangular piece was sawed from each corner at the junction of the circumference and the diametral saw line so that the halves would fit between the main columns during assembly.

The remaining parts of the machine, as well as fasteners mentioned previously, were available in the shop or were purchased. These items are listed in Table I.

This Brace Arm is Duplicated on
all Four Sides of Center Plate.

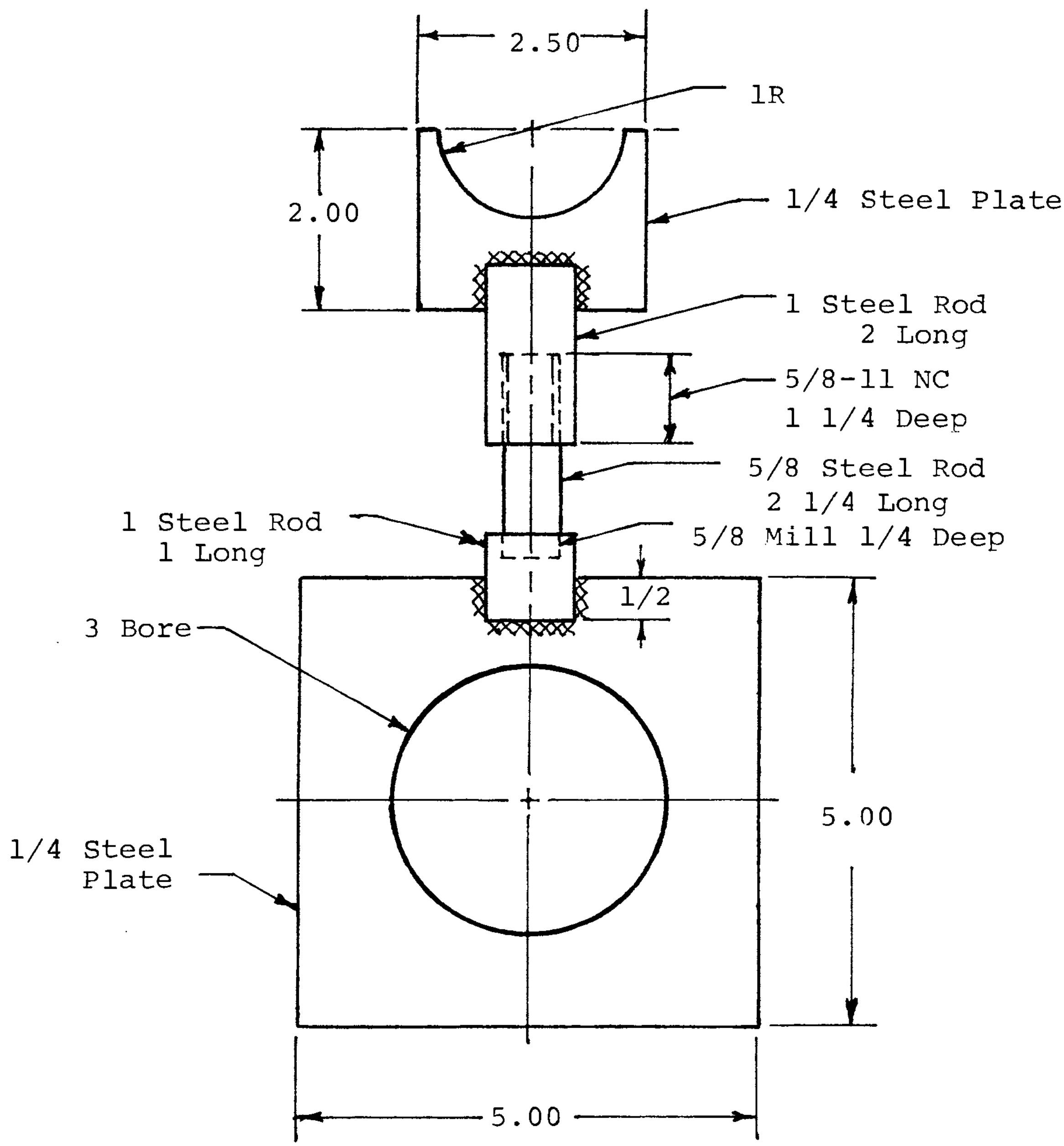


Figure 8. Brace Assembly

XXX Idicate Weld Both Sides
Tolerance ± 0.005

TABLE I
STANDARD PARTS

<u>Hydraulic Cylinder</u>

Miller Model 84H Heavy-Duty

3 1/4" Bore   6" Stroke

<u>Hydraulic Pump</u>

Enerpac Series III Model PM-742

Control Valve - 4 Way (Manual)

<u>Gage Adaptor</u>

Enerpac P-16.99

<u>Hi-Pressure Hose</u> (2 required)

Enerpac HC-914

6' with C-604 Coupling

<u>XX Steel Pipe</u> (2 required)

3/4" X 4" Threaded Both Ends

<u>Steel Couplings</u> (2 required)

3/4" 6000 psi

<u>Steel Bushing</u> (2 required)

3/4" X 3/8"

<u>Pressure Gage</u>

Blackhawk 0 to 10,000 psi

<u>Lathe Chucks</u>

Wescott #58 4-Jaw (2 required)

Skinner #4207-46 4-Jaw (1 required)

Bearings

Timken T-302-W Thrust (1 required)

New Departure #3211 (2 required)

Hex Nuts

1 1/2-12 NF (8 required)

1/2-20 NF (4 required)

5/16-18 NC (12 required)

Flat Washers

1 1/2	(8 required)
1	(2 required)
5/8	(12 required)
1/2	(4 required)
5/16	(28 required)

Lock Washers - Split-Ring

1 1/2	(8 required)
1/2	(4 required)
5/16	(8 required)

Set Screws

3/8-16 NC X 1 Half-Dog, Hex-Socket (8 required)

1/4-20 NC X 1 1/2 Cup Pt., Hex Socket (4 required)

Hex Head Cap Screws

5/16-24 NF X 3/4 (4 required)

5/16-24 NF X 1/2 (4 required)

Round Head Machine Screws

5/16-18 NC X 2 1/2 (4 required)

Square Nuts

5/16-18 NC (4 required)

U Clamps

5/16-18 NC X 2 X 3 1/2 (2 required)

Note: XX Steel Pipe is heavy-wall pipe testing 6,000 psi.

III. DISCUSSION

A. Set Up

The machine, prior to any changes, used a 1 horsepower, variable speed motor with a 2-inch diameter drive pulley to drive the flywheel by using the 4-inch diameter rear flange of the chuck as the driven pulley.

The first test was a no-load run to check for balance and vibration. There was some vibration observed at various motor speeds but was not found to be excessive, nor did the amplitude of vibration increase at any given speed. Because of this, the flywheel speed was limited to 1,000 revolutions per minute.

Following the initial testing of the machine, test specimens of 1-inch diameter were made; and, since the machine used 4 jaw chucks, flats were machined on the test specimens to provide a firm grip for the chuck jaws against the torque developed during welding.

The first runs were made at various angular velocities and pressures. It was evident from these runs that, with the angular velocity restricted to 1,000 revolutions per minute, it would not be possible to weld a specimen having a 1-inch diameter.

As a consequence, the specimen size was reduced to 1/2-inch diameter which could be successfully welded but which introduced a problem of axial alignment, shown in Figure 9. The applied pressures were sufficiently large and the chuck jaws loose enough to allow the two halves of the specimen to move during the initial contact portion of the welding cycle.

It was assumed that the unsupported extension of the hydraulic cylinder shaft was being misaligned by the torque forces exerted. To remedy this situation, supporting braces, Figure 8, were made and are shown in position in Figure 10. A number of runs were made with no significant improvement in the alignment of the welds.

A second attempt was then made by moving the braces closer to the weld zone and attaching them to the chuck jaws as shown in Figure 11. Again, there was no significant improvement in the weld alignment.

The chucks were then inspected, and it was concluded that, due to the large amount of previous use, the jaws should be moved toward the outer edge of the chuck, out of the worn area, and an adapter made to hold the specimen. The adapter would then be held in the chuck. This set-up is shown in Figure 12. A number of runs were made and it was noted that the chuck on the cylinder end was not holding properly. It was decided to do away with this chuck entirely and drill a hole in the ram adapter with four set screws to hold the specimen.

After several runs with this set-up, it was decided to replace the flywheel chuck with a heavier one to give more

Figure 9. Initial Specimens

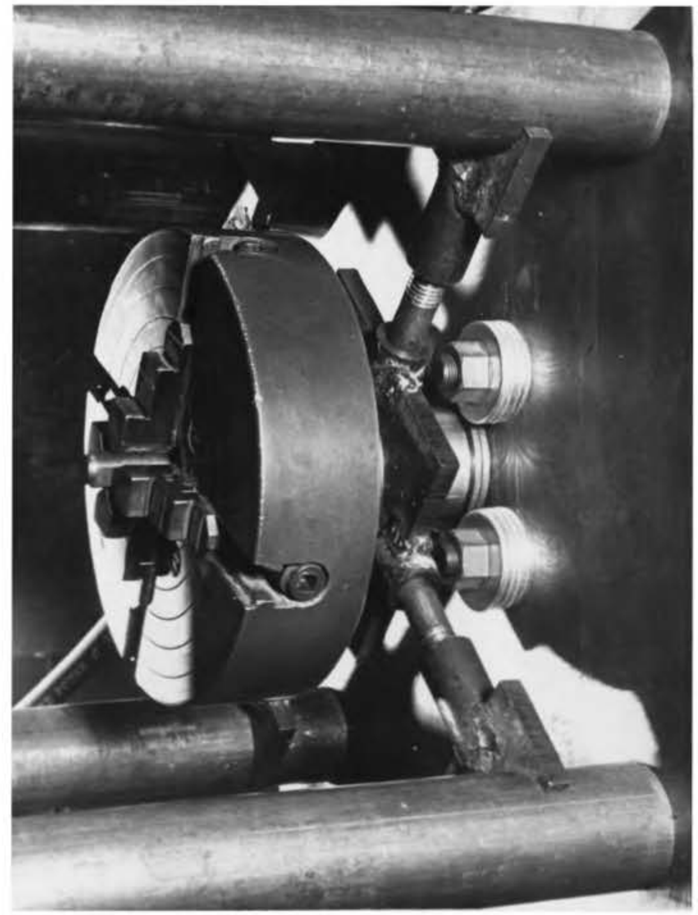


Figure 10. Second Welding Set-Up

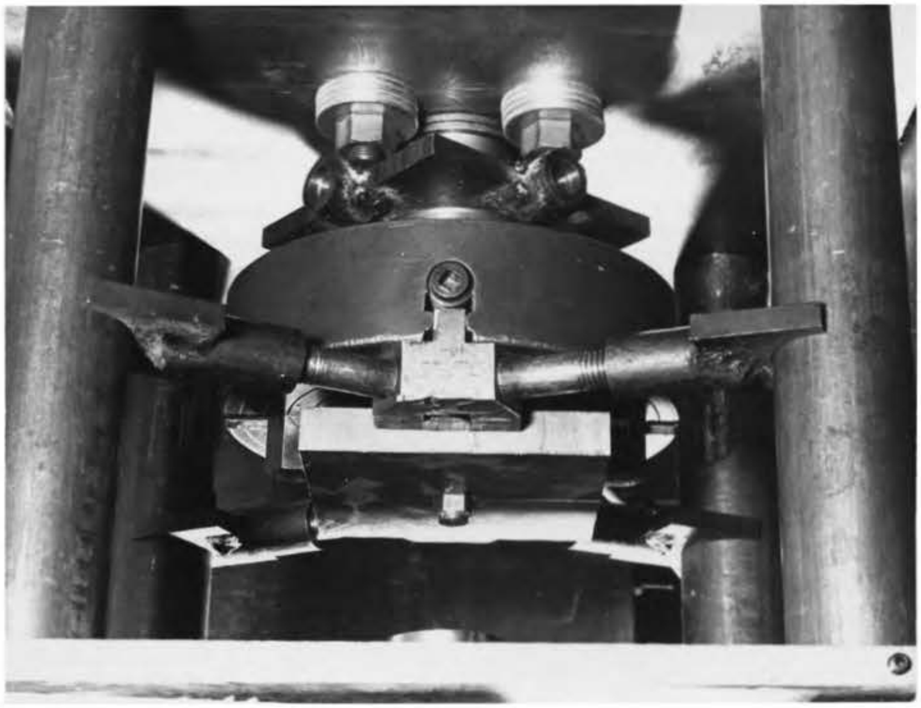


Figure 12. Fourth Welding Set-Up

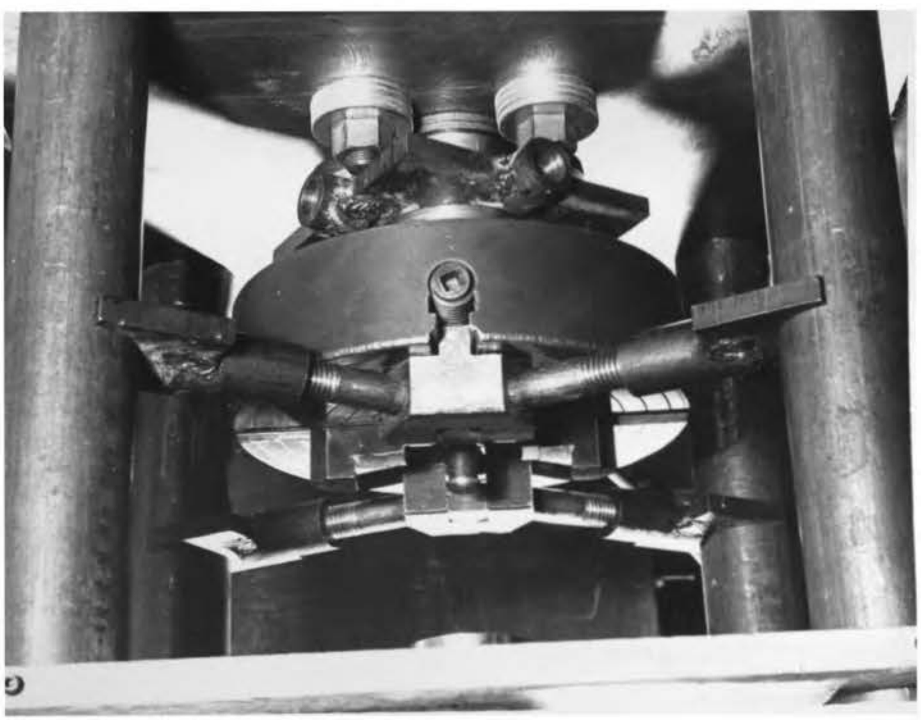


Figure 11. Third Welding Set-Up

rigidity to the specimen. There was still no significant change in the alignment of the finished weld, so this chuck was discarded also. An adapter was made to fit onto the fly-wheel shaft, as previously described, and this adapter was installed and tested. With this final set-up, the alignment problem was sufficiently corrected due to the fact that the specimen holder was now considerably more rigid and because the flywheel shaft support bearing was moved some 5 inches closer to the weld interface. This final set-up is shown in Figure 13, with a specimen ready to be welded, and in Figure 14 after the weld was complete.

The following were the steps involved in the set-up of the hydraulic system:

1. An Enerpac Series III Model PM-742 hydraulic pump fitted with a four-way manual valve was placed to the left of the machine.
2. A gage adapter was screwed into the right hand port and a 10,000 psi gage was screwed into the adapter.
3. 6-foot high-pressure hoses, with high-flow couplings, were screwed into both the adapter and the left valve port.
4. The hoses were then attached to a Miller Model 84-H hydraulic cylinder using high-pressure pipe fittings.
5. The pressure is adjusted by the use of an Allen wrench prior to making a run.

B. Test Procedure

The following steps were involved in making a run

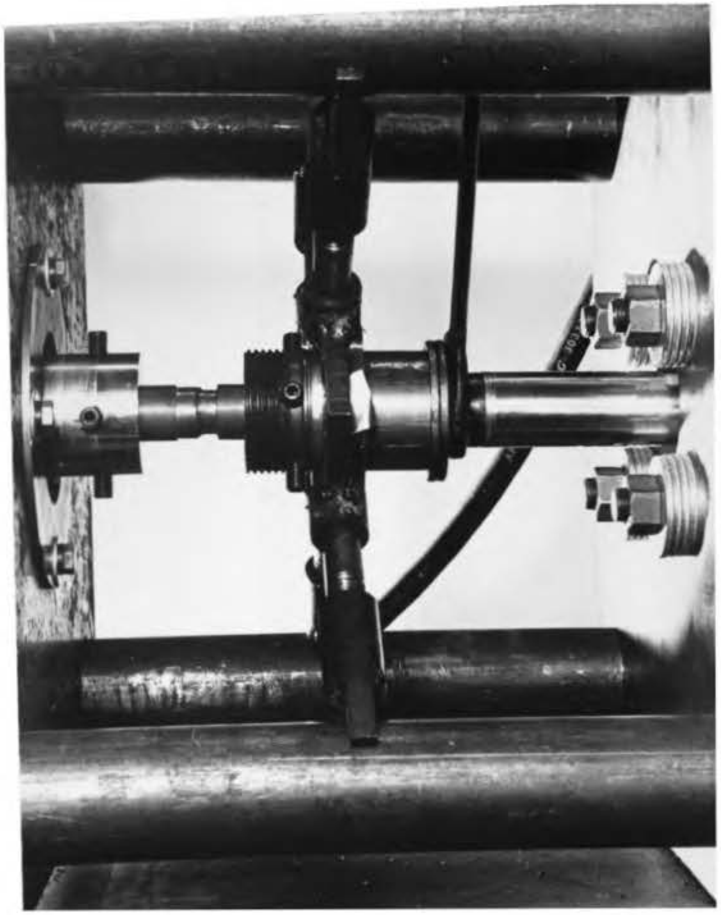


Figure 13. Final Welding Set-Up

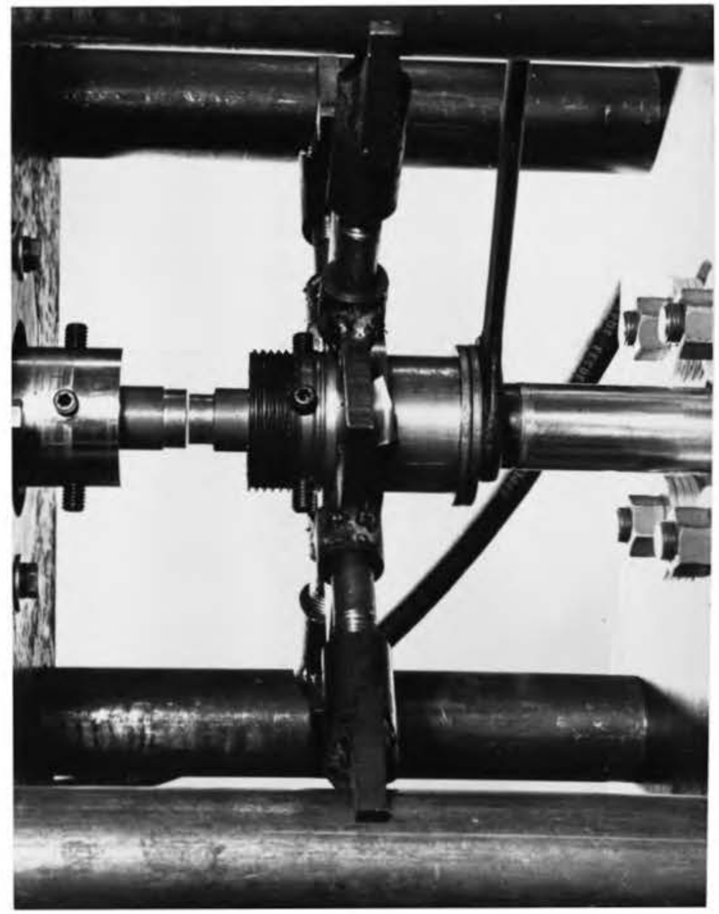


Figure 14. Specimen After Welding

and refer to the final set-up:

1. A specimen was placed in the flywheel specimen holder and the four set screws tightened against the flats on the specimen to prevent it from turning in the holder.
2. The same procedure as above was used to mount a specimen in the ram holder.
3. The ram was advanced until the specimen halves were approximately 1/32 inch apart.
4. The alignment braces were positioned and tightened.
5. The motor was started and the flywheel brought up to speed.
6. The motor was then cut off and simultaneously the valve to activate the ram was closed.
7. After the flywheel came to a stop, the valve was opened to relieve the axial force.
8. The braces were then removed.
9. The set screws were loosened to release the specimen.
10. The ram was withdrawn and the welded specimen removed.

C. Results

While making changes in the set-ups, no test data was taken other than to save a few specimens showing the alignment problem. These tests were run using 5/8" diameter rod turned down to 1/2" diameter on the ends. Several of these are shown in the photograph in Figure 7.

With the final set-up and the alignment problem corrected, data was taken to determine if the weld made fulfilled the requirements for an acceptable weld.

Run Numbers 1 through 17 were made with a 1/2-inch diameter solid specimen and a cross-sectional area of 0.196 square inches. It was determined after Run Number 3 that, due to an error in calculation, the axial force being used was creating stresses in the specimen much larger than the yield point stress of the material. This error was corrected in the following runs.

Different combinations of axial loads and angular velocities were tried until it was determined in Run 14 that the optimum conditions were achieved. Three more runs were made with the same parameters as Run 14. These specimens indicated that the angular velocity was too high since the flash was thrown out of the weld area rather than being squeezed out. To reduce the angular velocity would also reduce the amount of flywheel energy available for the weld and, due to the indications of previous runs, this would not give an acceptable weld. Two specimens, Run 14 and Run 16, were tested under tension and withstood 58,200 psi and 57,800 psi of stress respectively before fracture. The ultimate stress of the material is approximately 64,000 psi. This would be acceptable under normal conditions where design is based on the yield point stress and a safety factor considered as well. The yield point of the material welded was 40,000 psi.

Inspection of the fracture indicated a small spot in the center of the cross-sectional area that did not weld sufficiently, which was believed to be due to the fact that the exact center has zero velocity and, therefore, no heating due to friction. This area must receive heat by conduction from areas farther from the center. Since the process is so fast, the heat did not have time to transfer to this center area.

A hollow specimen would have no area of zero velocity, therefore, it was decided to use a hollow specimen but to retain the 0.196 square inch cross-sectional area. This was done for Runs 20 through 25. Run 26 was made using a specimen of 0.440 square inch cross-sectional area. The weld was made but the torque lifted the rear of the machine about 1/4 inch off the stand. No further runs with this larger area were made.

Runs 27 through 31 were made with specimens of 0.306 square inch cross-sectional area, and Specimens 28 through 31 were subjected to a tension test. Specimens 28 and 30 fractured at 65,450+ psi tensile stress. Specimen 29 fractured at 63,800 psi and Specimen 31 fractured at 60,800 psi of tensile stress. A test sample was taken from Specimen 27 and mounted and polished. Microscopic pictures were taken to show the grain structure at 100X, 250X, and 500X and gave excellent results. The weld zone shows no trace of an interface, but rather is a homogeneous small-grained structure showing no voids or irregularities.

Runs 27 through 31 were made with a flywheel weight of

200 pounds and an angular velocity of 800 revolutions per minute. The complete cycle took 1.0 second. The energy stored in the flywheel before starting the cycle was 2743 foot-pounds. The flywheel moment of inertia was 0.782 pound-feet-second^2. The horsepower per square inch of weld interface was 16.3 horsepower. Specimen 27 from which a sample was taken in order to make the microstructure photographs shown in Figures 5 and 6 is shown on the right in Figure 15. Also shown are both halves of specimen 30 after it fractured under a tensile stress of 65,450+ psi.

Figure 16 gives the entire welding set-up from the right front and left front, to show the components which make up the machine.

Figure 15. Test Specimens

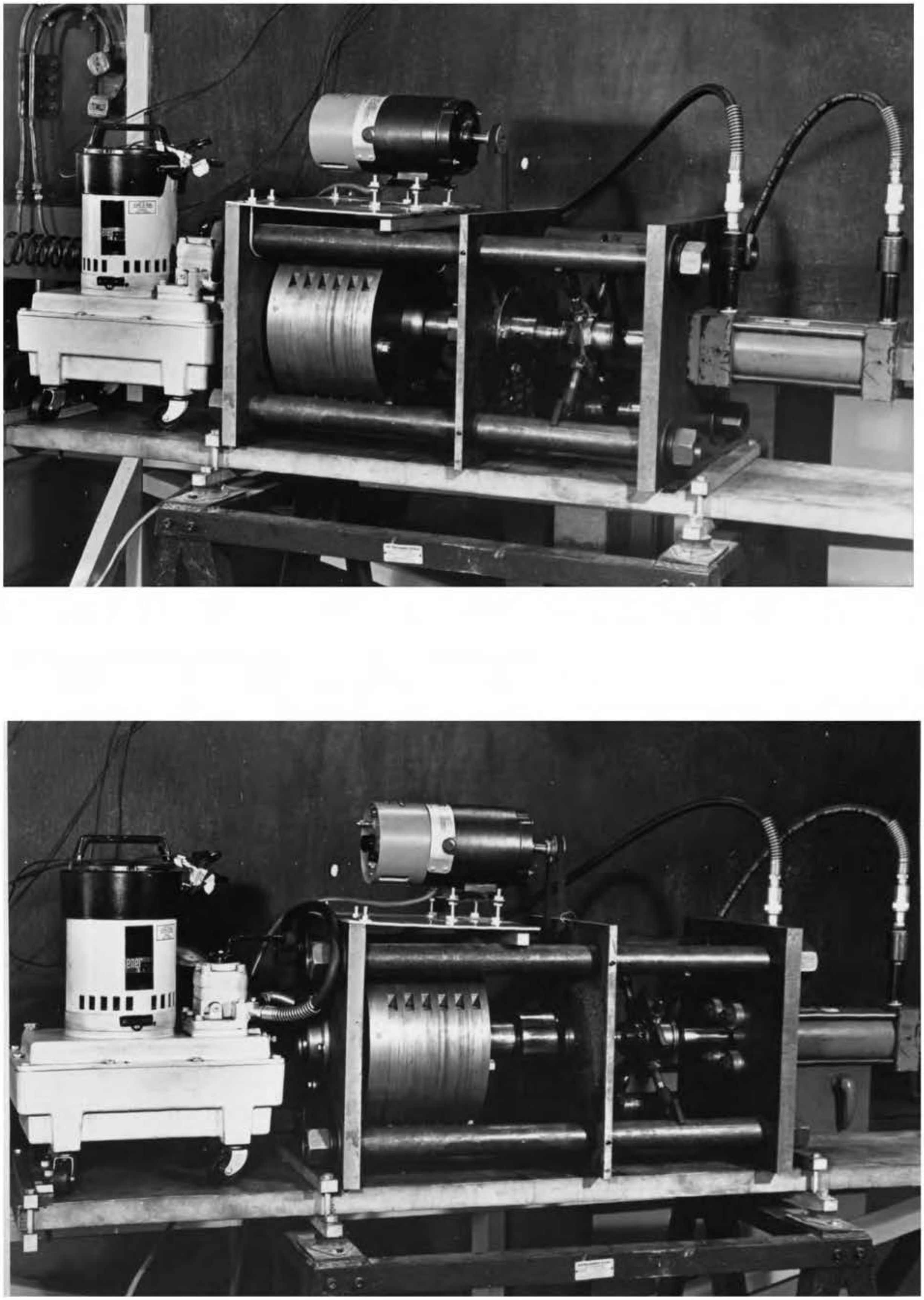


Figure 16. Entire Inertia Welder

IV. TEST DATA

TABLE II
1/2 INCH SOLID BAR

Run No.	Pressure psig	Thrust lbs.	Stress psi	Motor rpm	Remarks
1	-	-	-	-	
2	2,000	16,600	84,700	3,000	
3	2,000	16,600	84,700	2,500	
4	800	6,640	34,000	3,000	
5	800	6,640	34,000	2,500	
6	500	4,150	21,200	2,000	
7	500	4,150	21,200	2,500	
8	800	6,640	34,000	2,250	
9	800	6,640	34,000	2,250	
10	-	-	-	-	
11	500	4,150	21,200	2,500	
12	500	4,150	21,200	2,250	
13	600	4,980	25,400	2,250	
14	500	4,150	21,200	2,375	Fractured at 58,200 psi under tension
15	500	4,150	21,200	2,375	
16	500	4,150	21,200	2,375	
17	500	4,150	21,200	2,375	Fractured at 57,800 psi under tension

Note: Specimens used were of 0.196 square-inch cross-sectional area solid bar.

TABLE III
HOLLOW TUBES

Run No.	Pressure psig	Thrust lbs.	Stress psi	Motor rpm
20	600	4,980	25,400	1,500
21	600	4,980	25,400	2,000
22	750	6,230	31,750	2,000
23	750	6,230	31,750	2,250
24	650	5,400	27,500	2,250
25	550	4,560	23,200	2,250
26	1,100	9,150	22,600	2,500

Note: Specimens 20 through 25 were hollow with 0.196 square inch cross-sectional area in the wall. The outside diameter was 0.750 inches and the center hole of 0.562-inch diameter. Specimen 26 was of 0.440 square inch cross-sectional area. The outside diameter was 0.900 inches and the center hole of 0.500-inch diameter.

TABLE IV
HOLLOW TUBES

Run No.	Pressure psig	Thrust lbs.	Stress psi	Motor rpm	Tensile Strength psi
27	900	7,470	24,200	2,500	-
28	900	7,470	24,200	2,500	65,450+
29	900	7,470	24,200	2,500	63,800
30	900	7,470	24,200	2,500	65,450+
31	900	7,470	24,200	2,500	60,800

Note: These specimens were hollow with 0.306 square inch cross-sectional area in the wall. The outside diameter was of 0.800-inch diameter and the center hole of 0.500-inch diameter.

V. CONCLUSIONS

From the results obtained, it can be concluded that:

1. Imparting the same amount of energy to each weld for identical specimens gives results with a close tolerance.
2. The heat-affected zone is relatively small.
3. Heat requirements are lower for inertia welding than for friction welding.
4. Axial force required remains approximately that required to achieve 60% of the yield point stress for the material being welded.
5. Small diameter welds are increasingly harder to make due to the exceptionally high angular velocities needed to generate the necessary heat.
6. Welding of hollow parts requires less time and energy for a given cross-sectional area since no heat transfer to the center area is necessary.
7. Inertia welding gives greater uniformity than that obtained with friction welding.

VI. RECOMMENDATIONS

The following are the author's recommendations regarding this investigation.

1. Three-jaw, self-centering, heavy-duty chucks, operated either pneumatically or hydraulically, should be used.
2. The chucks should be enclosed in a bearing to keep them rigid during welding.
3. A more positive drive than that achieved with a single V-belt should be used.
4. Hollow spindles, onto which the chucks are attached, are preferable to facilitate the welding of specimens of longer length.
5. The flywheel should have a center of mass much farther from the axis of rotation than the 10 or so inches for the flywheel used in this investigation.
6. The mass of the flywheel should be easy to change to allow for different energy requirements.

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VIII. VITA

The author, Arthur Ferdinand Grimm, was born in St. Louis, Missouri, on April 25, 1928. He received his primary and secondary education at Wyman Grammar School and Roosevelt High School in St. Louis, Missouri. He received a Bachelor of Science Degree in Mechanical Engineering from the University of Missouri-Rolla in January, 1966.

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He worked for the Great Atlantic and Pacific Tea Company for 12 years prior to entering college and has also served in the United States Army for a period of two years.